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Analysis of recoverable exhaust energy from a light-duty gasoline engine

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ABSTRACT

While EER (Exhaust Energy Recovery) has been widely pursued for improving the total efficiency and reducing CO_2 emissions of internal combustion engines, the improvement on engine efficiency has been investigated with experimental work and numerical simulation based on a steam Rankine cycle EER system. The test was conducted on a light-duty gasoline engine connected with a multi-coil helical heat exchanger. Combining those experimental and modelling results, it demonstrates that the flow rate of working fluid plays a very important and complex role for controlling the steam outlet pressure and overheat degree. For achieving required overheat and steam pressure, the flow rate must be carefully regulated if the engine working condition changes. The flow rate has also significant influence on the heat exchanger efficiency. To achieving better heat transfer efficiency, the flow rate should be maintained as high as possible. From the simulation, it is found the EER system based on the light-duty test engine could increase the engine fuel conversion efficiency up to 14%, though under general vehicle operating conditions it was just between 3% and 8%. From the test, it is found the installation of heat exchanger can increase the exhaust back pressure slightly, the total fuel saving of the engine could be up to 34% under some operating condition.

1. Introduction

In recognition of the need to further reduce vehicle exhaust emissions and the greenhouse gas CO_2 as the oil price has kept roaring up, there has been an increasing interest in the development of cleaner and more efficient energy saving vehicle powertrain. It is thought future sustainable vehicle powertrain developments beyond the next decade are likely to be focused on four topics: emission legislation and control, new fuels, improved combustion and a range of advanced concepts for energy saving [1]. Among various advanced concepts, EER (Exhaust Energy Recovery) for Internal Combustion (IC) engines has been proved to not just bring measurable advantages for improving fuel consumption but also increase engine power output (power density) or downsizing, further reducing CO_2 and other harmful exhaust emissions correspondingly [2]. It was predicted by Vazaquez et al. that if 6% of the heat contained in the exhaust gases were converted to electric power, 10% reduction of fuel consumption can be achieved [3].

Early researches on EER have investigated the basic concepts, problems and expected improvements for such a system. An example could be found from the research conducted by Chammas and Clodic [4], who presented the advantages offered by a Rankine Cycle (RC) system designed for hybrid vehicles. Up to 18% fuel economy improvement could be achieved when water was used to recover the exhaust heat. Compared to the conventional exhaust waste heat recovery by turbocharger which has normally 15% fuel saving and significant power increase, Rankine cycle can have over 20% of fuel saving and similar power increase [5]. Although RC EER system has higher manufacture cost, it does not increase the exhaust back pressure obviously and it can be installed after turbocharger to regenerate further exhaust energy which turbocharger can't absorb.

Some recent reports have showed how further investigation of the technology and architectures are possible [6]. For instance, Teng et al. carried out a series of experiments [7–9] on heavy-duty diesel engines to explore the potential of EER, with hybrid energy systems combined the exhaust system with the charge air cooler and EGR cooler(s). Their results show that up to 20% increase in the engine power and 25% improvement in fuel savings over the ESC 13-mode test could be achieved by the EER system. Ringler et al. [10] selected two basic EER configurations (one just with exhaust gas only and another with exhaust gas plus coolant) from numerous illustrated Rankine cycle layouts for a detailed evaluation of heat recovery based on a four-cylinder IC engine. Their experimental works

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demonstrated that waste heat recovery can produce an additional power output of about 10% at typical highway cruising speeds. Weerasinghe et al. [5] identified the substantial potential of EER for IC engines via two most promising and technically viable technologies: turbo-compounding and exhaust heat secondary fluid power cycles. Their results revealed that the two EER technologies would contribute more power output in the order of 4.1%–7.8% and fuel savings by 2%–22%. Various researches have underlined the interest in light to heavy-duty vehicle applications and suggest that fuel economy improvements of over 20% can be expected from EER.

In this paper, the study which has been focused on the exhaust energy from a light-duty gasoline engine with the objective of exploring the available recoverable energy in exhaust gas is presented. While the exhaust temperature and gas flow rate vary with engine operating conditions, the available exhaust energy for a steam Rankine cycle EER system and its characteristic under different engine operating conditions would be understood. Then the optimal operating areas for utilizing the exhaust energy could be identified.

2. Parameter definition and model description

The EER system which will be used in the present research is based on Rankine cycle and physically it comprises four main components: an evaporator/heat exchanger, an expander, a condenser and a pump, as shown in Fig. 1.

With the evaporator/heat changer, the working fluid is superheated by absorbing thermal energy provided from the exhaust gas. Flowing out from the evaporator as high temperature steam, the working fluid is driving the expander to produce useful work. Then the waste steam from the expander will be cooled down through the condenser to return to liquid phase. In the next step, the working fluid is pumped to maintain the circulation.

For most internal combustion engines, there is approximately 20–40% of total fuel energy which is dissipated through exhaust gas, with the majority as sensible enthalpy due to high exhaust temperature and the minority as chemical enthalpy due to incomplete combustion. To evaluate energy amount in the exhaust gas, it is necessary to obtain the thermo-physical parameters of exhaust gases. Considering currently all diesel engines and most gasoline engines during dominant operating period are driven with lean combustion condition, an assumption of complete in-cylinder combustion would be used for the following analysis while the focus of this research work is on the thermal energy recovery of exhaust gas. Then small amounts of incomplete combustion products such as CO and unburnt hydrocarbon and other small emission components such as NOx could be ignored and the compositions of exhaust gases could be considered as the mixture of CO_2, H_2O, N_2 and O_2.

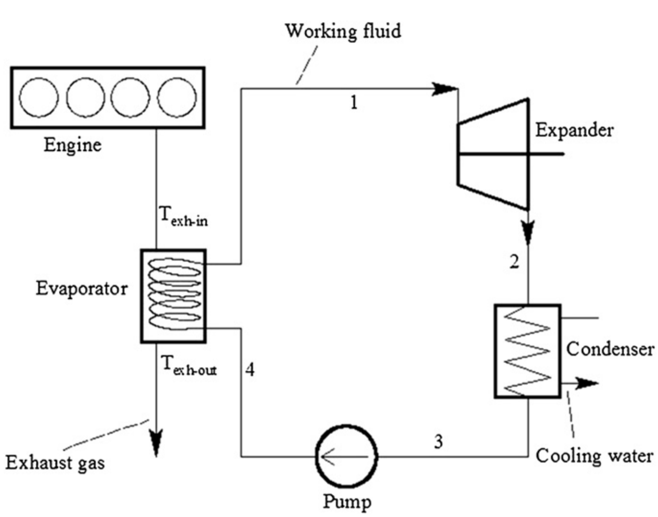

Fig. 1. Layout of the EER system.

Provided that the stoichiometric air-fuel ratio in the gasoline combustion is a_0, and the actual one is α, the molar fractions for N_2 and O_2 in the air mixture is k_{N_2} and k_{O_2}, respectively. While the atom numbers of carbon and hydrogen in the hydrocarbon fuel molecular are θ_C and θ_H, respectively, the molar fractions of compositions in the exhaust gases could be obtained by the following equations, respectively.

$$\begin{aligned}
\varphi_{N_2} &= \frac{ak_{N_2}}{1+a+a_0k_{O_2}-\theta_H/2} \\
\varphi_{O_2} &= \frac{(a-a_0)k_{O_2}}{1+a+a_0k_{O2}-\theta_H/2} \\
\varphi_{CO_2} &= \frac{\theta_C}{1+a+a_0k_{O_2}-\theta_H/2} \\
\varphi_{H_2O} &= \frac{\theta_H/2}{1+a+a_0k_{O_2}-\theta_H/2}
\end{aligned} \tag{1}$$

Considering the above four compositions are all ideal gases, their constant pressure heat capacity $C_{p,i}$ could be achieved by the empirical formula [11]:

$$C_{p,i} = \left(c_0+c_1T+c_2T^2+c_3T^3+c_4T^4\right)R \tag{2}$$

Given the exhaust gases ideal condition, the specific enthalpy could be calculated by:

$$h = \sum_i \omega_1 M_i h_i \times 10^{-3} \tag{3}$$

where, ω_i, M_i and h_i are the molar fraction, molar mass and specific enthalpy for each composition. And the latter could be expressed by:

$$h_i = h_0 + \int_{T_0}^{T} C_{p,i}\mathrm{d}T \tag{4}$$

Combined the equations (1)–(4), the specific enthalpy of exhaust gases can be achieved.

It should be noted the above formulae can only be selected for calculating the specific enthalpy of exhaust gases when all thermal recovery process did not involve steam condensation of exhaust gas and there is only the sensible heat of the exhaust gases which is absorbed by the thermal recovery system. When the steam condensing heat should be included if there is phase change of exhaust gas via the evaporator, the exhaust specific enthalpy was obtained from NIST-Refprop database.

Then, the exhaust heat q_{exh} and its fraction in the total fuel energy could be given by:

$$q_{exh} = (h_{exh}-h_{out})\dot{m}_{exh} \tag{5}$$

$$\chi_{exh} = \frac{q_{exh}}{h_f\dot{m}_f} \tag{6}$$

where, $\dot{m}_{exh}$ and $\dot{m}_f$ are the mass flow rate of exhaust gas and fuel, respectively, h_{exh} and h_f are the corresponding exhaust gas enthalpies, and the low heating value of fuel.

After possible sensible heat in the exhaust gas could be determined with above formulae, the EER Rankine cycle efficiency would

be estimated by considering those several critical parts in the cycle, including evaporator, expander, condenser and pump.

In the present research, a helical coiled heat exchanger as showed in Fig. 1 is used as the evaporator. The heat transfer coefficient in the evaporator was calculated by including the forced convection between the hot exhaust gas and the external surface of the coiled pipe, the conduction in the coiled pipe wall and the forced convection between the internal surface of the coiled pipe and the working fluid (liquid water or/and steam). The total heat transfer coefficient in the evaporator can be expressed as:

$$\frac{1}{h_{ev}} = \frac{1}{h_1}+\frac{1}{h_2}+\frac{1}{h_3} \tag{7}$$

where, h_{ev} is the total evaporator coefficient, h_1, h_2, h_3 are the heat transfer coefficient between the hot exhaust gas and the external surface of the coiled pipe, in the coiled pipe wall and between the internal surface of the coiled pipe and the working liquid or the steam, respectively.

For estimating the heat transfer coefficient between the exhaust gas and the external surface of the coiled pipe, the Colburn correlation [12] for the external flow of force convection is used as the following:

$$\mathrm{Nu} = 0.036\mathrm{Re}_L^{4/5}\mathrm{Pr}^{1/3} \tag{8}$$

For the heat transfer between the internal surface of the coil pipe and the working liquid or steam, Dittus-Boelter correlation [13] for the internal turbulent flow of forced convection is employed.

$$\mathrm{Nu} = 0.023\mathrm{Re}^{0.8}\mathrm{Pr}^{0.4} \tag{9}$$

Although in the coiled pipe it exists two-phase flow during heat transfer process, this part was neglected in the current simulation due to its complexity. In the next section for the comparison between simulation results and test data, it can be found the influence for neglecting the two-phase flow during heat transfer stage is very small on the heat transfer results.

The total rate of heat transfer and the evaporator efficiency are:

$$q_{ev} = \dot{m}_{WF}h_{ev}A\Delta T_{LMTD} \tag{10}$$

$$\eta_{ev} = \frac{q_{ev}}{q_{exh}} \tag{11}$$

where A is the surface area of the coiled pipe. ΔT_{LMTD} is the log mean temperature different which can be calculated as [13]:

$$\Delta T_{LMTD} = \frac{\Delta T_{exh} - \Delta T_{WF}}{\ln\frac{\Delta T_{exh}}{\Delta T_{WF}}} \tag{12}$$

where ΔT_{exh} is the temperature difference of exhaust gas between the exchanger inlet and outlet, ΔT_{WF} is the temperature difference of working fluid between the outlet and the inlet.

The efficiency in the expander was determined with the turbine data provided by the turbine supplier. When the working pressure ratio is about 1.5–2.5 bar, the efficiency ranges from 55%–65%. Then, the flow condition at the outlet of the expander can be decided as:

$$q_{ev}\eta_{exp} = \dot{m}_{WF}\left(h_{exp-in} - h_{exp-out}\right) \tag{13}$$

In the condenser, the heat lost can be expressed with the following formula. The outlet temperature of the condenser is assumed to be controlled as optimal condition, such as around 90–95 °C for water.

$$q_{cond} = \dot{m}_{WF}\left(h_{exp-out} - h_{cond-out}\right) \tag{14}$$

The work consumed by the pump can be estimated as:

$$w_{pump} = \frac{\dot{m}_{WF}}{\rho_{WF}}\left(P_{pump-out} - P_{pump-in}\right) \tag{15}$$

Then the EER Rankine cycle efficiency can be calculated as:

$$\eta_{RC} = \frac{q_{ev}\eta_{exp} - w_{pump}}{q_{exh}} \tag{16}$$

The fraction of the net output work from EER Rankine cycle in the total fuel energy is:

$$\chi_{EER} = \chi_{exh}\eta_{RC} \tag{17}$$

This fraction is considered as the EER efficiency in the current research when the energy lost from the expander to the energy final use was not taken into account.

3. Test description

As a close loop of EER Rankine cycle system is still under construction, the present experiment is conducted with an open thermal cycle system which is connected with a light-duty gasoline engine. In Fig. 2, it shows the main parts and instrumentations of the experimental system. It can be seen the evaporator is a helical coiled heat exchanger which is supplied the working fluid by a water tank via a water pump and a flow rate control valve. In the present research, the working fluid used for the open-loop EER system is hot water which was maintained around 80 °C for those tests demonstrated in this paper. The flow parameters of working fluid are collected by a flow rate meter, two thermocouples for inlet and outlet temperature and two pressure sensors for inlet and outlet pressure.

The test engine used is a 1.3 L gasoline engine for which main relevant specifications are listed in Table 1. The heat exchanger used as the evaporator is connected on the exhaust pipe 0.4 m downstream from the 3-way catalyst. Exhaust temperatures before and after the heat exchanger and the back pressures before exchanger are measured accordingly. The exhaust gas flow rate is collected from the engine air mass flow meter.

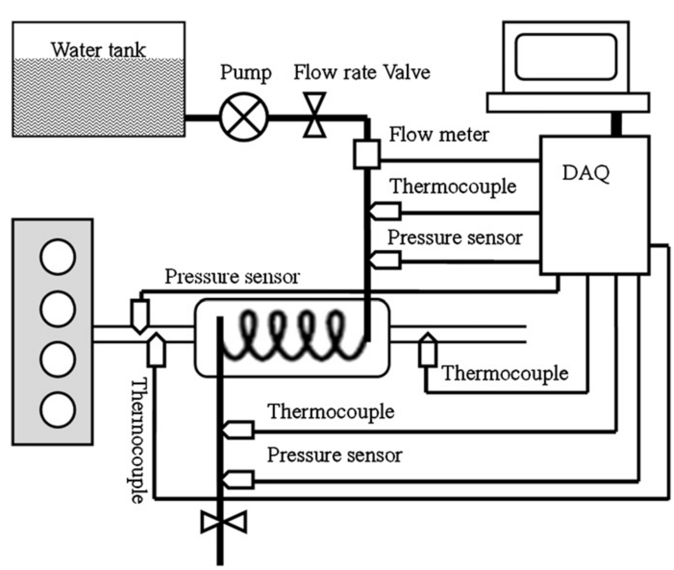

Fig. 2. Experimental setup of the open-loop EER system based on a gasoline engine.

Table 1
Main specifications of the test engine.

Engine type	CA4GA1
Number of cylinders	4
Bore × stroke (mm)	73 × 80
Displacement (L)	1.339
Compression ratio	10
Number of valves	16
Rated power/speed (kW/rpm)	67/6000
Maximum torque/speed (Nm/rpm)	120/4200

For achieving adequate heat exchange efficiently, the evaporator of EER system is a multi-coil helical pipe heat exchanger which is shown in Fig. 3. The specifications of the exchanger can be found in Table 2.

Table 2
Specifications of helically coiled heat exchanger.

Outer shell inner diameter (mm)	130
Inner shell outer diameter (mm)	38
Shell wall thickness (mm)	1.5
Shell material	304 stainless steel
Tube outer diameter (mm)	8
Tube wall thickness (mm)	0.8
Tube material	316 stainless steel
1st coil diameter (mm) (from inner to outer)	49
2nd coil diameter (mm)	71
3rd coil diameter (mm)	93
4th coil diameter (mm)	115
Main length for heat transfer (mm)	700
Overall length of heat exchanger (mm)	920

4. Results and discussion

4.1. Energy characteristics in exhaust gas

For analysis of exhaust energy, several parameters of air mass flow rate and exhaust gases, such as temperature, mass flow rate, would be required for the calculation of exhaust energy. In those parameters, the exhaust temperature plays a so important role, such as for the design of recovery system, the choice of working fluids and the optimization of system. Therefore, in order to make full advantages of exhaust energy, it is necessary to have adequate information of the distribution of exhaust gases temperature under different engine operating conditions. As shown in Fig. 4, it is the measured exhaust gas temperature as function of engine speed and corrected engine load (torque).

It could be found that the exhaust gases temperature depends strongly on the engine speeds and loads. Here it should be noted the actual vehicle operation normally needs the engine speed ranges from 2000 to 4000 rpm. During the mid-range of the engine load (40–80 Nm), the corresponding exhaust gases temperature of the test engine can be 500 °C–700 °C, while it will be up to 850 °C at the full load. It should be pointed out the data in Fig. 4 were measured in the outlet of 3-way catalyst, which is followed downstream by the heat exchanger of EER system. Therefore, based on the exhaust temperate distribution, an appropriate evaporating temperature and mass flow rate would be determined for the working fluids in the thermodynamic cycle of exhaust energy recovery.

Based the exhaust temperature distribution shown in Fig. 4, the sensible heat in the exhaust gas can be estimated by considering the gas flow rate and the engine operating air-fuel ratio. As shown in Fig. 5, it is the distribution of the ratio of exhaust sensible heat to the total fuel energy. The highest ratio can be up to 40% located around the highest power area in the map, and the lowest is around 20% located at lower torque area in the.

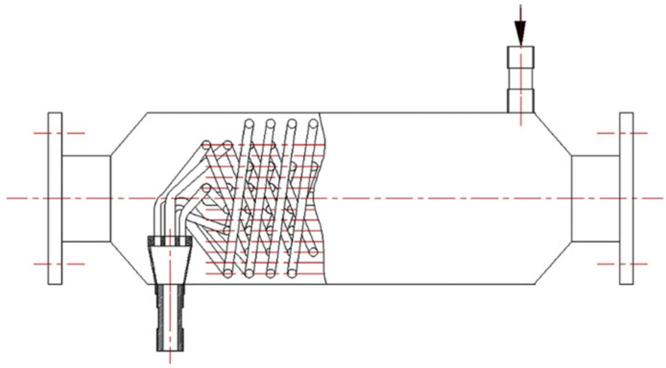

Fig. 3. Evaporator/heat exchange.

When the engine efficiency improvement via EER is assessment, it is necessary to find the possible energy which can be recovered by the EER system. Then the ratio of exhaust sensible heat to the total fuel energy will provide possible information for designing optimal EER system which can provide the best performance under required engine operating condition.

4.2. Heat exchange test result and model validation

When the exhaust gas condition of the test engine was investigated, the working fluid parameters before and after the heat exchanger were measured by the flow meter, thermocouples and pressure sensors in the open-loop thermal cycle system. During those tests, the flow rate of the working fluid was controlled by the control valve. The outlet pressure was regulated by the regulating valve, then the overheat can be adjusted to the required valve. As shown in Fig. 6, they are the variations of the control pressure as function of the flow rate under different overheat value. Those points marked with symbols are experimental results and those lines are plotted from the simulation data with the model introduced in Section 2. Those tests were carried out under constant engine speed of 2000 rpm and constant torque of 32 Nm.

From the results, it can be seen, both results from experiment and simulation are in very good agreement. Although those low overheat degrees are probably not suitable for the expander, the results shown in Fig. 6 provide useful validation for the simulation model, in particular the heat exchanger model which plays a very important role in the simulation.

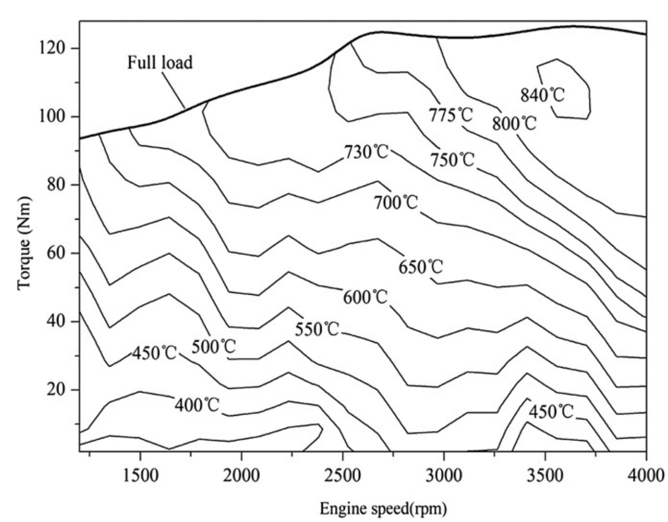

Fig. 4. Distribution of exhaust gases temperature of the test engine as function of engine speed and torque.

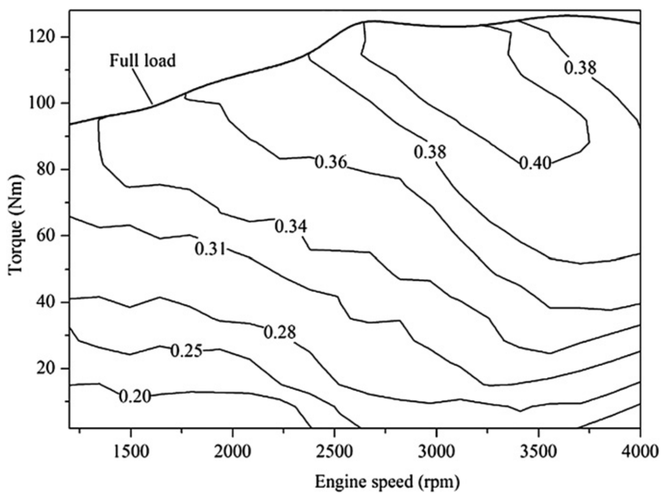

Fig. 5. Distribution of the ratio of exhaust sensible heat to the total fuel energy.

From the test and simulation results, it can be found that the flow rate must be reduced for increasing the working pressure under constant overheat. It suggests when the exhaust gas condition has some variation, a pump controller system must work to control the flow rate when required overheat and the exchanger outlet pressure need to be maintained.

Based on the above measurement data, the evaporator efficiency was estimated and the results are shown in Fig. 7. Those curves are still simulation results. It can be seen the evaporator efficiency is dependent on the flow rate of the working fluid very much, but is almost not affected by the overheat degree. This demonstrates when the priority is given to the overheat degree or/and the output steam pressure of the heat exchanger, there is possible loss or increase on the evaporator efficiency due to the variation of the flow rate of the working fluid.

4.3. Analysis of EER system performance

Based on the validated model, analysis of the EER system performance was conducted. When the efficiency of the expander and work losses in the pump of Rankine cycle are considered, the whole system efficiency were simulated under the above engine operating condition (2000 rpm and 32 Nm) with just 100 K overheat. As shown in Fig. 8, it can be seen the whole EER system efficiency can be over 11% when the flow rate is about 6.4 kg/h. The efficiency can decline to less 7% when the flow rate is reduced to 5.4 kg/h.

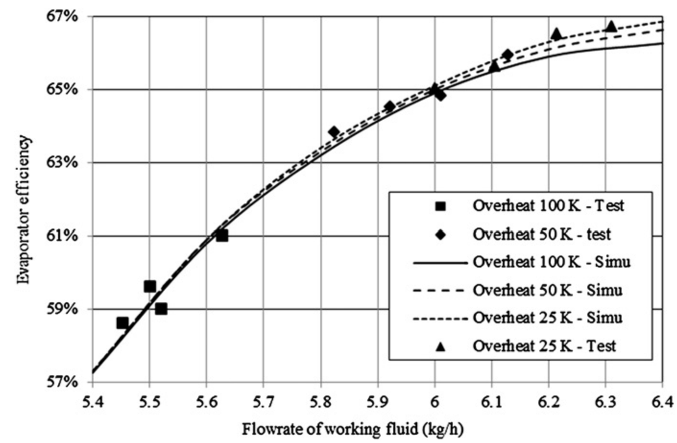

Fig. 7. Evaporator efficiency as function of the flow rate of working fluid.

Still under 100 K overheat, the optimal system efficiency under different engine speed and engine load was analysed and the result was plotted in Fig. 9. From the plot, it can be seen that the best EER system efficiency for the light-duty test engine can be up to 14%. However, this requires the engine operating condition must close the peak power area. In the area where passenger car engines general work, the efficiency is around 3% and 8%.

4.4. Influence of EER system and engine performance

As presented in Section 3, the heat exchanger has 4 coils which in one hand is for providing enough heat transfer, in the other hand may increase back pressure of the exhaust. For exploring the influence of the heat exchanger on the back pressure then on the fuel consumption, the test was carried out under constant engine speed of 2000 rpm. Shown in Fig. 10, it can be find the exchanger caused a slight higher back pressure then some increase of fuel consumption.

Compared to the increases on the exhaust back pressure and the engine fuel consumption due to the installed of EER heat exchanger, the improvement on the total engine efficiently is analysed and the

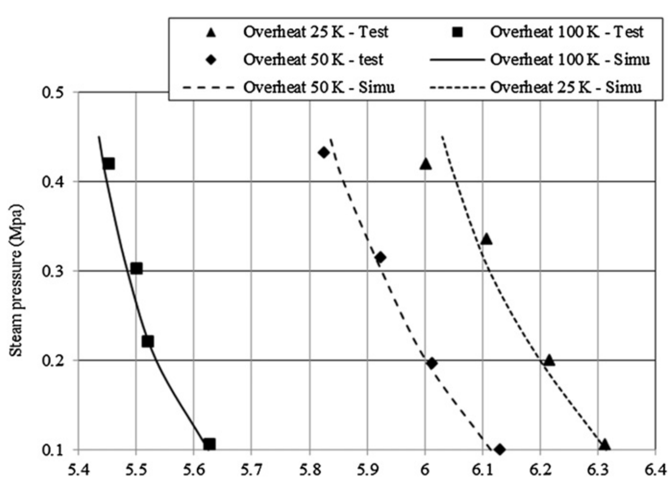

Fig. 6. Experimental and modelling results of variations of steam pressure as function of flow rate under different overheat.

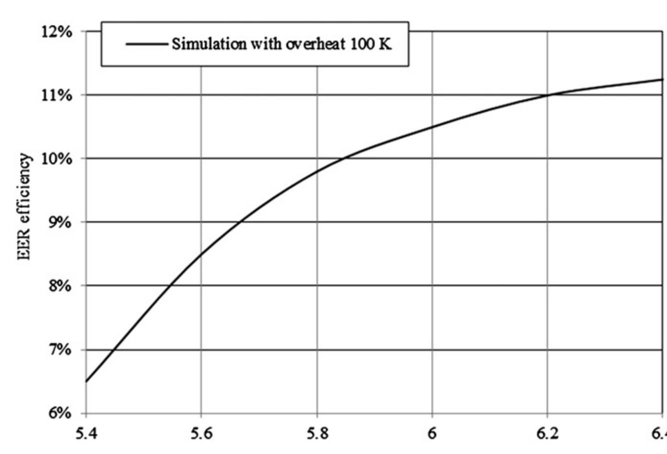

Fig. 8. Variation of the whole EER system efficiency as function of the flow rate of working fluid.

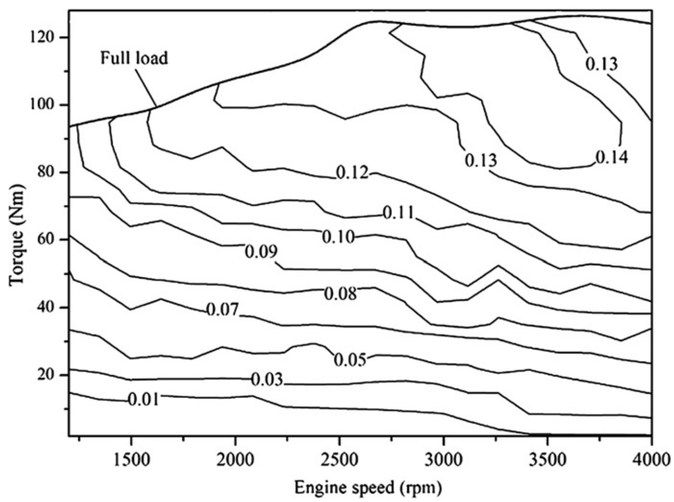

Fig. 9. EER system efficiency distribution as function of engine speed and load.

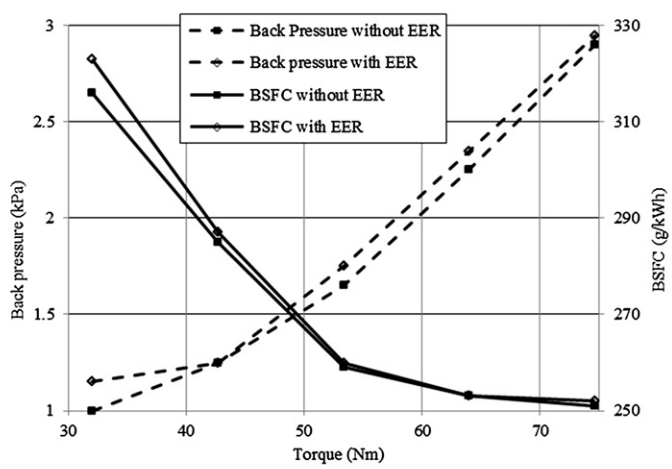

Fig. 10. Influence of the heat exchanger on the exhaust back pressure and the engine fuel consumption.

result is shown in Fig. 11. It can be seen, under 2000 rpm, the total engine efficiency (fuel conversion efficiency) could increase to the maximum value of 44% with EER from the maximum value of 32.5% without EER. The increase rate which is about 34% in theory means

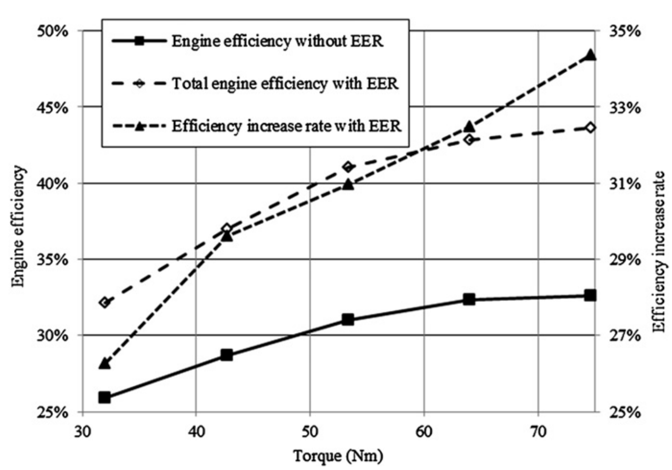

Fig. 11. Engine (fuel conversion) efficiency increase with EER system.

the fuel saving could be up to 34% due to the installation of EER system.

5. Conclusions

In the current study, experiments were performed on a light-duty gasoline engine for obtaining thermodynamic parameters of exhaust gases under different operating conditions. Based on the experimental data, the exhaust heat and maximum recoverable exhaust energy for an EER (Exhaust Energy Recovery) system based on steam Rankine Cycle were calculated and analysed, including the characteristics in terms of temperature and other operating conditions. From those experimental and simulation results, the following conclusions can be derived:

- The flow rate of working fluid plays a very important and complex role for controlling the steam outlet pressure and overheat degree. For achieving required overheat and steam pressure, the flow rate must be carefully adjusted if the engine working condition changes.
- The flow rate has also significant influence on the heat exchanger efficiency. To achieving better heat transfer efficiency, the flow rate should be as high as possible.
- From the simulation, it is found the EER system based on the light-duty test engine could increase the engine fuel conversion efficiency up to 14%, though under general vehicle operating conditions it was just between 3% and 8%.
- Although the heat exchanger can increase the exhaust back pressure slightly, the total fuel saving could be up to 34% under 2000 rpm and 75 Nm.

Acknowledgements

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